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W. Zhou  
*University of Cincinnati*

J. Kim  
*University of Cincinnati*

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Zhou, W. and Kim, J., "Prediction of the Noise Radiation of Hermetic Compressors Utilizing the Compressor Simulation Program and FEMIBEM Analyses" (1996). *International Compressor Engineering Conference*. Paper 1168.  
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# **PREDICTION OF THE NOISE RADIATION OF HERMETIC COMPRESSORS UTILIZING THE COMPRESSOR SIMULATION PROGRAM AND FEM/BEM ANALYSES**

W. Zhou and J. Kim

Structural Dynamics Research Laboratory  
Department of Mechanical, Industrial and Nuclear Engineering  
University of Cincinnati  
Cincinnati, OH 45221-0072, USA

## **ABSTRACT**

The complexity arising from the interactions between the pulsating flow, gas cavity and shell structure has been the major obstacle in realizing the prediction of the radiated sound level of hermetic compressors by a numerical simulation. In this work, a numerical procedure combining a computer simulation program of the compressor process and the Finite Element Method (FEM)/Boundary Element Method (BEM) techniques is used to demonstrate the feasibility of such predictions. The simulation program is used to calculate the unsteady flow inputs to the inside shell cavity of the compressor. A FEM/BEM approach is used to calculate the resulting shell motion and the radiated sound field.

## **INTRODUCTION**

Shell cavity gas pulsation has been recognized as a major contributor to the total noise level of hermetic compressors [1]. Therefore, numerical prediction of the noise radiation from the hermetic compressor shell resulting from the gas pulsation inside its cavity has been desired for a long time. Numerous works based on analytic and numerical [2, 3] or experimental [1, 4] approaches can be found related to this task. However, no complete numerical simulation of the radiated noise of a hermetic compressor starting from its source mechanism has been reported.

If the radiated noise induced by the pulsating flow in the shell cavity can be estimated with reasonable accuracy by a purely numerical method, it would be very useful for compressor designers. For example, it will allow a truly optimal suction muffler to be designed because it can be designed based on concrete data, not by a guesswork. Currently all necessary tools required for such a simulation are available. For example, four pole theories to be applied to three dimensional systems [2], FEM/BEM analysis software and compressor simulation programs are all available. Therefore, major obstacles related to this task are considered as the lack of a systematic procedure because of the complexity of the problem. It is the purpose of this paper to demonstrate that a numerical estimation of the shell noise radiation by fully considering all interaction aspects is feasible.

In this work, a compressor simulation program is utilized to calculate the spectrum of the pulsating flow in the shell cavity at the suction muffler inlet of a low side reciprocating compressor. Then, the four pole method is applied to calculate the pulsating flow induced at the suction return pipe point according to the procedure developed in Reference [2]. Pressure response functions which are necessary to calculate the four poles are obtained by a BEM analysis. The natural modes of the shell are calculated by a FEM analysis and imported to the BEM model. Finally, the noise level radiated to the external free field is calculated by a BEM analysis. The geometry of the compressor used in this study was somewhat simplified, however, retaining most of its important characteristics. The results demonstrate that it is certainly possible to evaluate the sound radiation of hermetic compressors by a purely numerical simulation procedure.

## FEM/BEM MODELING AND ANALYSIS

A slightly simplified geometry of the compressor shell and cavity was considered. The acoustic cavity bounded by two cylinders of different lengths which form an annular cavity whose top and bottom are open is shown in Figure 1. The compressor shell is represented by the outside cylinder and the internal part of the compressor is by the inside cylinder whose walls are assumed to be rigid. Two different geometries of the shell top, one flat and the other curved, were considered to observe the influence of the shell geometry on the sound radiation. The gas in the cavity has the sound speed of 170 m/s and the density of 5.17 kg/m<sup>3</sup> which approximately corresponds to the properties of R134a at 50°C. The effect of the lubrication oil was not considered in this study.

The size of the cavity and the location of the suction muffler inlet and return pipe outlet are shown in Figure 1. The thickness of the shell, 1.5 mm, was intentionally taken thinner than typical shells used in small compressors to see stronger structural interaction effect.

As the first step, the finite element method is used to obtain modal data of the shell and the acoustic cavity. The first eight modes of the shell and cavity calculated for both geometries are listed in the Table 1. As expected, the modal responses of the two types of the shell are quite different from each other due to the curvature effect of the top. The natural frequencies of the flat top shell are much smaller than the corresponding ones of the curved shell. However, there is no significant change in the acoustic modes of the two cavities, which suggests that minor geometric changes without accompanying the volume change would not make large differences in acoustic modes.

Shell					Cavity		
Mode No.	Flat Top		Curved Top		Mode No.	Flat	Curved
	Freq (Hz)	Comments	Freq (Hz)	Comments		Freq (Hz)	Freq (Hz)
1	975	top mode	1815	side mode	1	371	395
2	1023	top mode	1815	side mode	2	460	457
3	1687	side mode	1868	side mode	3	460	457
4	1687	side mode	1868	side mode	4	667	686
5	1695	side mode	2124	side mode	5	667	686
6	1695	side mode	2124	side mode	6	758	774
7	1928	top mode	2347	side mode	7	836	833
8	1928	top mode	2347	side mode	8	836	833

Table 1. Natural Frequencies of the Shell and the Cavity

## CALCULATION OF PULSATING FLOW SOURCES

For a low side compressor, the input flow to shell cavity is the flow induced by the pumping action of the compressor at the inlet point of the suction muffler. The flow strength can be calculated in frequency components by the computer simulation program as shown in Figure 2. Then, the four pole method is used to calculate the source strength at the return pipe of the suction line. The sources in the cavity are modeled as monopoles because their sizes are much smaller compared to the size of the cavity and the wavelength in the frequency of interest.

The relationship between input source strength  $Q_1$ , acoustic pressure  $p_1$  and output source strength  $Q_2$ , acoustic pressure  $p_2$  can be expressed as:

$$\begin{Bmatrix} Q_1 \\ p_1 \end{Bmatrix} = \begin{bmatrix} A & B \\ C & D \end{bmatrix} \begin{Bmatrix} Q_2 \\ p_2 \end{Bmatrix} \quad (1)$$

where, A, B, C, D are the four pole parameters [5], which can be calculated as follows [2]:

$$A = -\frac{a_{22}}{a_{21}}, B = \frac{1}{a_{21}}, C = a_{12} - \frac{a_{11}a_{22}}{a_{21}}, D = \frac{a_{11}}{a_{21}} \quad (2)$$

where,  $a_{ij}$ ,  $i, j = 1, 2$ , is the pressure response function which is defined as the pressure response at point  $i$  due to the unit source input at point  $j$ . The values of these pressure response functions at a specific frequency are obtained by a BEM analysis. Because the BEM result gives an infinite pressure response at the monopole source point due to its related formulation procedure, the pressure response functions at source points  $a_{11}$  and  $a_{22}$  were calculated at a point located at the distance of the half of the radius of the pipe. A further study to clarify this problem is in progress. Applying the anechoic end condition at the output side in Equation (1), the induced source strength  $Q_2$  was calculated.

### PREDICTION OF SOUND RADIATION

Radiated noise levels were calculated at the frequencies of 176, 352, 410 and 997 Hz. Except the first, all frequencies were chosen around the natural frequencies of the lowest acoustic or shell modes (see Table 1).

The source strengths at these frequencies are listed in Table 2. Notice that the sources  $Q_1$  and  $Q_2$  are the flows at the muffler inlet and suction return point, and the latter is calculated by applying the four pole method. Because of the phase difference,  $Q_2$  becomes a complex number.

Freq (Hz)	Flat Top		Curved Top	
	$Q_1$ (m <sup>3</sup> /s)	$Q_2$ (m <sup>3</sup> /s)	$Q_1$ (m <sup>3</sup> /s)	$Q_2$ (m <sup>3</sup> /s)
176	0.3909e-03	-0.5005e-08+j0.1005e-06	0.3909e-03	-0.3145e-08+j0.6261e-07
352	0.3201e-04	-0.3562e-07+j0.2825e-06	---	---
410	---	---	0.1510e-04	-0.6265e-08+j0.6324e-07
997	0.2025e-06	0.9956e-09-j0.3532e-09	0.2025e-06	-0.1301e-09+j0.4366e-08

Table 2. Source Strengths in the Acoustic Cavity

Because the indirect BEM of the commercial software used in this work could not handle multi-media problem, the calculation was made using one medium (R134a) for both the inside and outside of the shell. Then, the exterior sound pressure level (SPL) was adjusted by the following equation:

$$SPL_{air} = 20 \log \left( \frac{P}{P_{ref} (\rho_{air} / \rho_{R134a})} \right) \quad (3)$$

where,  $P$  is the calculated external pressure,  $\rho_{air}$  and  $\rho_{R134a}$  are the densities of air and refrigerant, to obtain the SPL in air. The justification of this practice or an alternative method is also in progress in the author's current work.

### DISCUSSION OF THE RESULTS

Figures 4 to 8 show the polar plots of the radiated sound predicted along two circles at 1 meter from the center of the compressor. Figure 3 shows these two circles for the data recovery and the coordinate system. The zero degree line in the plots coincides with the positive x-axis or y-axis. Figure 4 shows the radiated sound level at 176 Hz plotted along the circle on the xz plane. Because the lowest non-

zero natural frequencies of the shell and the cavity are far higher than 176 Hz, the radiated noise shows no directivity. The yz plane plot which is not included also didn't show any directivity. Therefore, a constant strength sound field is induced at this frequency. The estimated SPL of about 52 dB at this frequency corresponds to 40 dBA which may have been over estimated since the shell is so thin.

Figure 5 and 6 show the comparisons of the SPL at the frequencies of 352 Hz and 410 Hz which are close to the first acoustic natural frequencies of the respective cavities. The overall SPL from the flat top shell is higher than that of the curved top shell as expected.

Figure 7 and 8 show the SPL at 997 Hz which is close to the lowest natural frequency of the flat top shell. As expected, SPL radiated from the flat top shell is much higher than that of curved top shell, particularly around the top of the compressor.

### CONCLUSION

It was demonstrated that the noise radiation from a hermetic refrigeration compressor could be calculated by a purely numerical procedure. Although there remain a few issues to be addressed further, such as handling of the singularity and effect of the cooling oil, it is believed that the results are accurate enough to be used for practical purposes. This work is considered to be the first numerical implementation to obtain the sound pressure level of hermetic compressors considering the cylinder process, gas pulsations, and shell vibrations. The best application will be found in the design of the muffler system because it can be designed based on the actual SPL rather than indirect information such as transfer functions.

Future works to improve or supplement this procedure are considered as:

- Including the lubrication oil,
- Refinement of the model geometry,
- Handling the singularity of the pressure response function at the source point,
- Direct way to handle multi-media system.

Some of the above issues are already in progress in the current work being done by the authors.

### ACKNOWLEDGMENT

The support for the first author by Samsung Electronics Co. Ltd. for the work related to this paper is greatly acknowledged. The provision of Comet/Acoustic BEM/FEM analysis software by Automated Analysis Corporation is also acknowledged.

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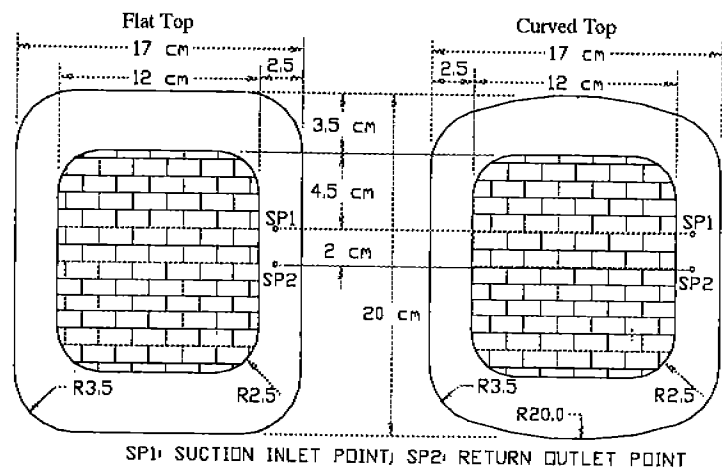


Figure 1. Dimensions of the Compressor

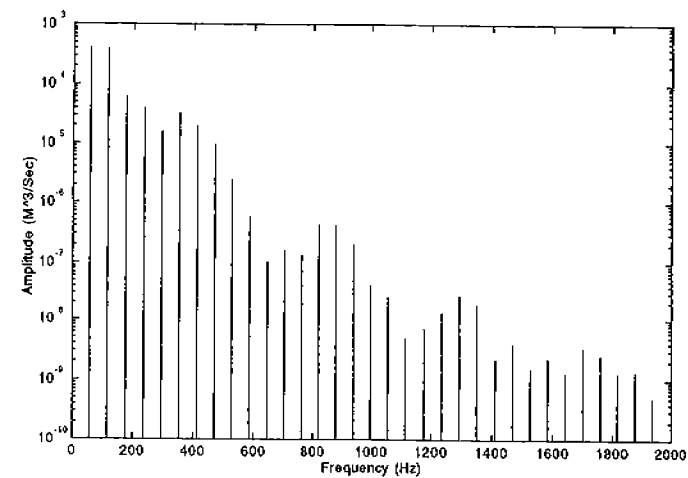


Figure 2. Frequency Components of the Input Source

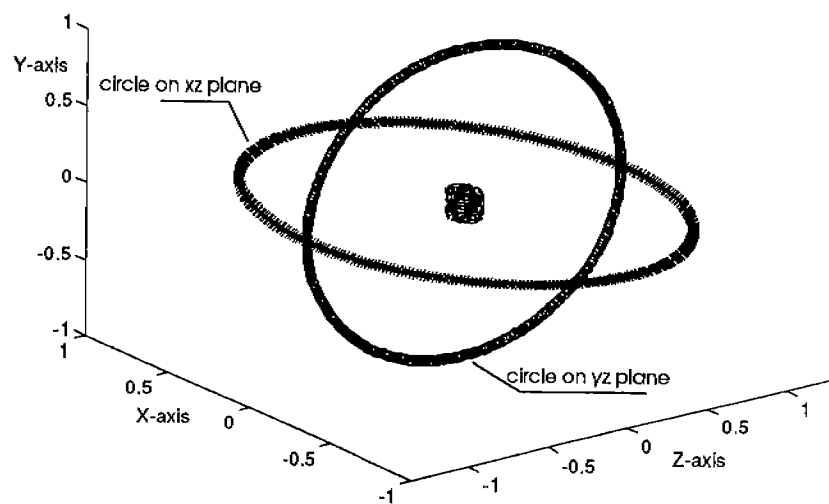


Figure 3. Circles for the Polar Plots

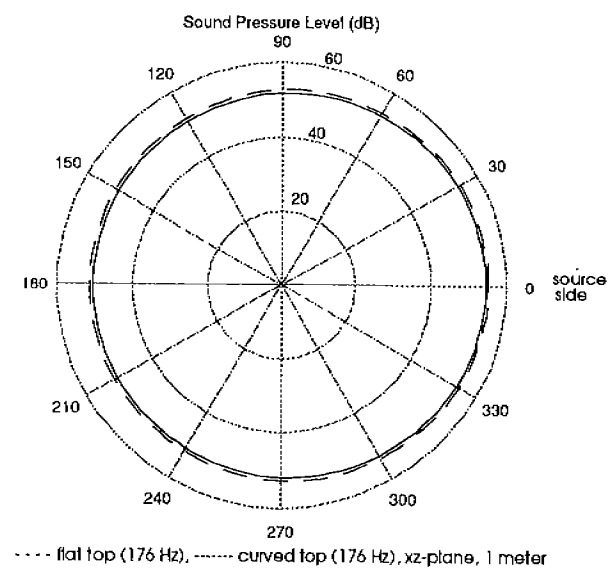


Figure 4

